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ENERGY MINIMISATION CRITERION AS A METHODOLOGICAL GUIDE TO THE DYNAMIC ANALYSIS AND OPTIMISATION OF FULLY TRIMMED AND EQUIPPED VEHICLES

Francesca Avenati - Bassi*[1], Sara Lorea[1], and Francesco Carlo Tinti[2]

[1]Rieter Automotive Systems S.p.A., Strada del Fornacino 119, 10040 Leinì – TO, ITALY
[2]Rieter Automotive Systems S.p.A., Via G. Agnesi 251, 20033 Desio - MI, ITALY
francesca.avenati@rieterauto.com

Abstract

Customer demand for increased vibro-acoustical comfort performances of vehicles, coupled with reduced development time frame, requires a continuous improvement process in the engineering design phase.

These needs call for fully comprehensive FEM models, in order to simulate vehicle performances already in the very early stages of the project: for this purpose, a big amount of detailed information is necessary, as the virtual vehicle should be as close as possible to reality and thus complete of trim and equipment. Moreover, being the first target setting and final verification phases of the development process experimentally based, a good numerical-experimental correlation is a milestone.

The validated numerical model should be used in an efficient and systematic way in order to outline criticalities, find-out the important transmission paths and optimise the performances: an innovative approach for this goal has been conceived, focussed on energy distributions and minimisation criteria as a guideline in the selection of crucial structural improvements.

The paper fully describes the complete methodology, both in its theoretical aspects and practical issues, as it was applied during a development of a new D-segment car for the emerging Asian market. The virtual vehicle has been realised either by direct or equivalent dynamic modelling and provided with the whole dynamic information coming from the technical specifications or the experimental database. The acoustic cavities have been modelled with their real geometric shapes and volumes and complete of phono-absorbing properties of the different trimmings. Both the local behaviour and the vibro-acoustic coupling of the structure with the internal cavities have been analysed and the newly-conceived methodology has been applied in order to solve different types of criticalities.

INTRODUCTION

The development of new vehicles suffers nowadays from two almost opposite needs: namely, the growing importance of the vibro-acoustic performance as an added value and the reduced time frame of the project. While the first implies an increasing amount of data and analysis, the second necessarily shortens the experimental phase on prototypes calling for a “virtual vehicle” to rely on.

In this paper, the development of “virtual dynamic” vehicle (D-segment car for the Asian market) is described as derived from the opposite tasks of completeness and limited dimensions; as a validation, its performances are compared with measurement results on the predecessor. The simulation activity procedure is described in details with a special focus on a newly conceived procedure for structural analysis and countermeasures selection.

PROJECT OVERVIEW

Methodology

The preliminary, but fundamental, phase in the project development has been to create a „Virtual dynamic“ vehicle, reproducing the completely assembled one in real measurement conditions (i.e. fully trimmed and with all dynamic systems): its performances have then been compared with measurements carried out on benchmarking vehicles.

In details, the following curves have been chosen as guiding parameters in the vibro-acoustical design of the vehicle: the inertances (a / F) at the attachment points of major vibrating sources to the body and the pressure transfer functions (p / F) to the trimmed passenger cavity. Being in fact these attachments the first “gate” for the vibrations coming from the sources, the a/F curves represent the best physical indicator of the body structure stiffness properties: a lack in them implies an easy way for the vibrations to enter into the vehicle. At the same time, the p / F curves contain a direct indication of the goodness of the coupling between the structure and the air cavity: the peaks in these transfer functions are connected either to an high level of vibration of panels facing the passenger compartment or to an eigenmode of the cavity.

The project has been subdivided into two phases, in which the model has been successively increased in complexity and validated by direct comparison with measurements on the predecessor vehicle. The analysis sessions have been organised in the following steps: after the location of critical frequencies and attachment points, the major modal contributors to the response function have been extracted (implying, for the p / F case, the investigation on both the trimmed panels and the fluid cavity contributions).

Then, a methodological approach has been established for the structural optimisation: kinetic and strain energy distributions have been evaluated near the analysed attachments and structural modifications have been defined according to energy type mainly contributing. Finally, the improvements on the response spectrum have been verified on the optimised model, by using it in a predictive way.

Virtual dynamic vehicle assembly and model validation

All the structural parts were implemented into the "body in white" model, together with systems, equipment and trimmings in order to be as close as possible to reality. Either an equivalent system or the complete F.E. modelling of the parts has been chosen, according to their impact to the whole dynamic behaviour, thus resulting in reduced model dimensions.

The amount of technical information needed was enlarged, the dynamic behaviour strongly depends, for example, on the dynamic stiffness of the mountings, as well as on the damping properties of the different components. As the technical specification values often quite differ from measured performances, an experimental database to access to is a must. As a model validation, a good correlation has been achieved with respect to measurements on the predecessor car, thus allowing for the further analysis steps (see Fig. 1).

Special care has been paid to the fluid-structure interaction modelling: as the fluid modes are strongly affected by the geometric shape of the trimmed passenger cavity, the occupied volumes (e.g. seats, IP, tunnel console) have been removed; moreover, externally simulated absorption performances of the different trimmings have been introduced and special investigations have been performed in analysing the trunk cavity impact on the final p / F results. The numerical-experimental correlation found, even if partially influenced by a different acoustic treatment concept, allowed for the further analysis process (see Fig. 2).

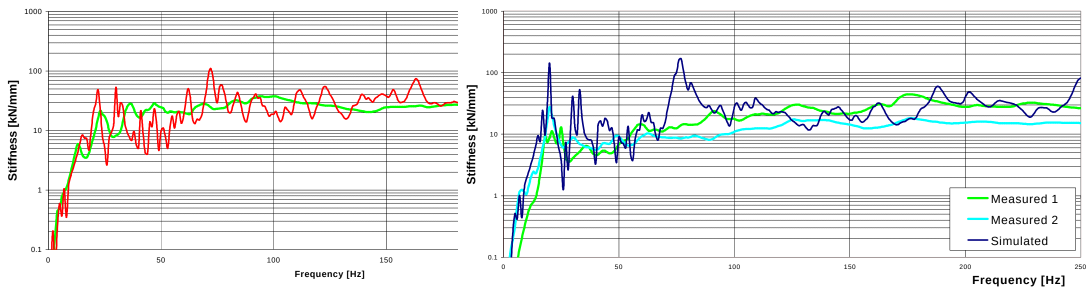

Figure 1 - numerical / experimental correlation: a / F transfer functions.

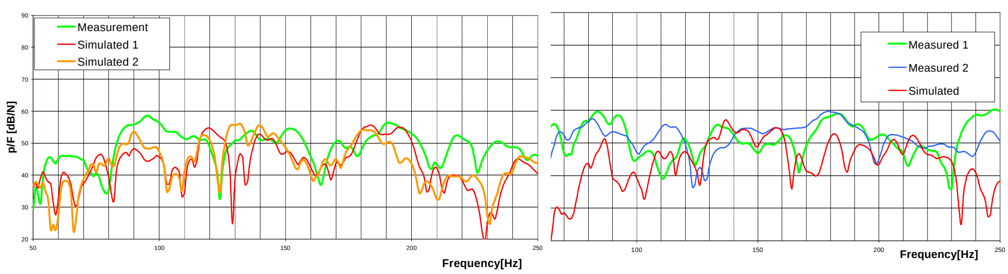

Figure 2 - numerical / experimental correlation: p / F transfer functions

Vibro-acoustic simulation details

(a / F) and (p / F) have been simulated for frequencies up to 250 Hz by means of MSC Nastran (SOL111) solver, by applying unit input forces at the attachments of the following sources: powertrain, exhaust, front and rear suspension braces. The (p / F) curves have been evaluated at two different points inside the passenger cavity (centre front and rear mikes).

Vibration performance analysis and model dimension reduction

The simulated a / F have been transformed into dynamic stiffness curves and compared with specific target curves, previously defined by a benchmarking measurement activity. Critical frequency ranges have been located by direct inspection of the simulated stiffness.

The problem dimensions have been reduced by extracting the “modal participation factors” and focussing on the most contributing modes.

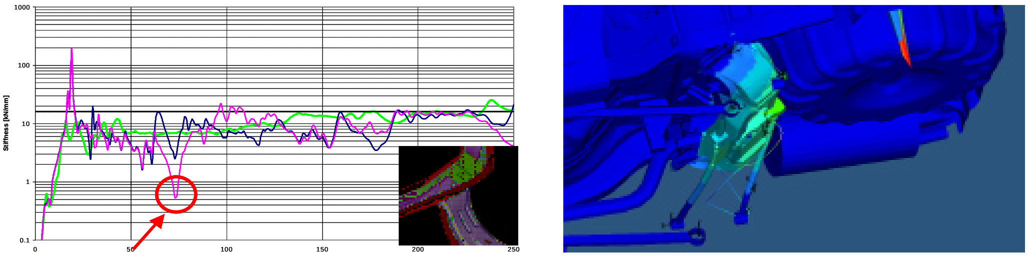

Figure 3 - dynamic problem in the body structure at rear subframe attachment points

Vehicle vibration performance optimisation: energy minimisation criteria as leading strategy

The classical approach to structural performance optimisation deals with the direct inspection of the previously extracted modal shapes: looking at their animated display, major modes can be subdivided into either “global” (which could be hardly improved without direct intervention on the structural architecture) and “local”, usually cured by locating maxima in the strain energy. However, this process isn’t controlled in final result, being the intervention mainly based on the engineering experience and “sense” of the designer. Moreover, a second type of possible actions can be foreseen in the optimisation process of a dynamic phenomenon: namely, tuned dynamic resonators and added masses which can’t be actually designed according to the “static strategy” as they’re driven by the kinetic energy.

The optimisation process here described has been carried out by looking at the sum of kinetic and strain energies at each finite element.

The interventions have been methodologically selected by extracting energy distributions around the located critical attachment: for strain energy dominating, stiffening actions were chosen, otherwise masses and resonators were designed.

Energy distribution analysis and statistics

Firstly, the extracted element's energies have been directly visualised on the structure (colour-maps) and compared with the contributing mode-shapes. Together with practical sense and experience, this approach could be enough to define the correct intervention, especially when either strain or kinetic energy is clearly dominating (see Fig. 4, left part).

However, a more systematic approach has been established focussing on the near-by of the interested attachment and performing a statistical analysis of the sample. As maximum values could have been misleading or affected by the model meshing characteristics, percentage distributions of the different energy types have been extracted and graphically represented: e.g., for the rear subframe attachment (see Fig. 4, right part), the relevance of strain energy is evident, as the behaviour of the total one well copies the strain curve both in its positioning along the energy axis and in the percentage values.

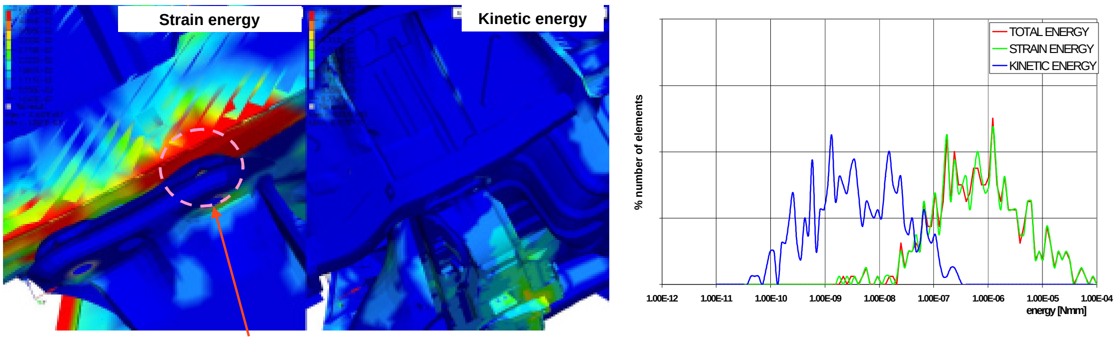

Figure 4 - rear subframe attachment points: maps and percentage energy distributions

Statistical parameters able to fully describe the phenomenon have been found by investigating all the different attachments: the Gaussian formulation appeared the more suitable distribution to fit the data, and the mean value, standard deviation and the "probable error" (P.E.) have been chosen as guiding parameters. While the meaning of the first two is straightforward, as they allow for the energy ranking (for each sample of elements the one characterised by the higher mean and lower standard deviation has been identified as "dominant"), the last parameter has been introduced for the most complicated cases.

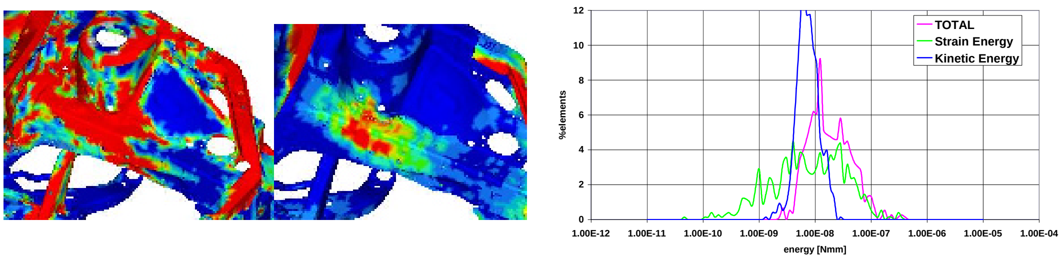

Figure 5 – gearbox side attachment points: maps and percentage distributions

It may happen that both strain and kinetic energy distributions are characterized by similar mean values, but one of them is sharper and peaked (see Fig. 5): according to the common sense, it is the dominating one. The P.E. parameter (defined as half the interval, around the mean value, containing 50 % of the element energy values) is the correct indicator: lower P.E. values correspond to higher peaks around the mean conjugated with a lower spread. As an example, comparing values in Table 1 and 2, it can be noticed that, while for rear-subframe attachments the mean is enough for pointing out the strain as dominant energy, for the gearbox point the lower P.E. value represents a better parameter to rank the kinetic energy as dominant.

Table 1 - rear subframe attachments: statistical parameters [N mm]

Energy type	Mean Value	Standard deviation	P.E.
Strain	6.71E-07	8.17E-01	5.51E-01
Kinetic	3.91E-09	8.26E-01	5.57E-01

Table 2 – gearbox side attachments: statistical parameters [N mm]

Energy type	Mean Value	Standard deviation	P.E.
Strain	7.56E-09	6.46E-01	4.36E-01
Kinetic	7.36E-09	1.99E-01	1.34E-01

Once the major energy has been identified, the corresponding intervention has been designed and applied to the original model; a simulation session has been carried out resulting in new (a / F) curves, element's strain and kinetic energies.

Both the improvement on the analysed (a / F) curve as well as the pertained energy reduction could be verified by comparison with the original data: e.g. the originally dominant strain energy had to be significantly decreased in the optimised model, implying a total energy reduction and a / F enhancement.

In Fig. 6 and 7 the enhancement on the stiffness curves and the corresponding decrement on energy distributions are graphically represented: in the first case, the model modification involved the introduction of a reinforcement on the structure (strain energy was dominating), while in the second a tuned dynamic resonator has been designed (kinetic energy mainly contributing).

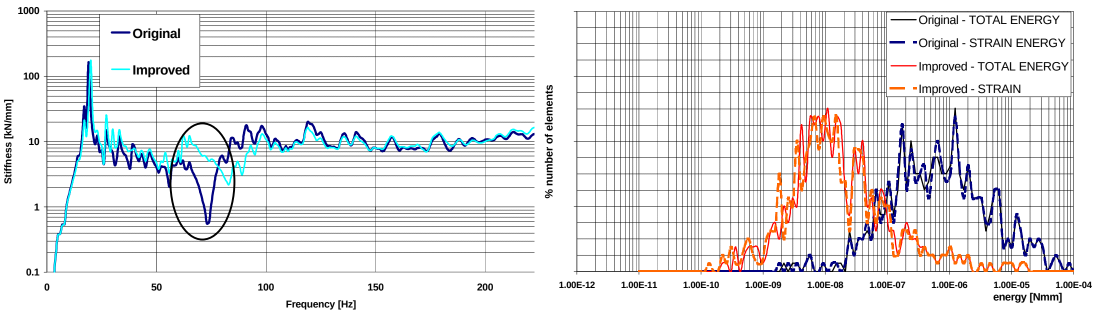

Figure 6 – rear subframe attachment points: improvement on stiffness curve and corresponding energy distribution decrement

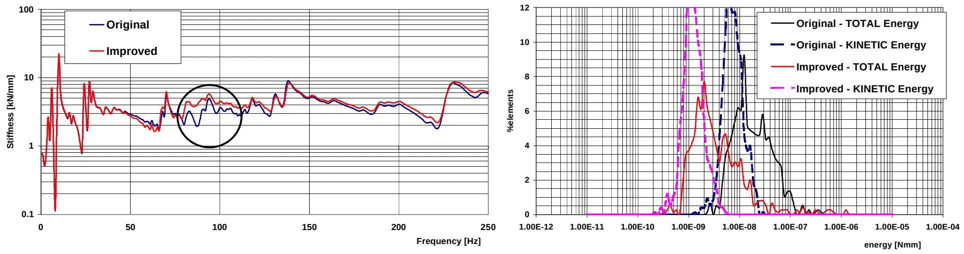

Figure 7 – gearbox attachment points: improvement on stiffness curve and energy distributions

Optimisation process - Analysis of results

The established optimisation process has been applied to several cases, resulting in a good correspondence between dominant energy types, successive a / F improvement and total energy reduction. Moreover, counter-checks have been performed by designing interventions not in line with the energy distribution statistics: they resulted in either lower or almost null decrements in the a / F peaks and in slight energy variations. As an example, the energy distributions at the engine side attachment pointed out the kinetic energy as dominant (see Fig. 8), thus leading to the design of a tuned resonator: as a countercheck, also a structural reinforcement has been defined and introduced into a second model. In Fig. 9 the comparison between the two modified models is represented: while the resonator significantly improved the response curve, no modification has been produced by the reinforcement; consistently, the total energy has been decreased only in the first case, as the other one affected only the non-dominating part of the global energy content.

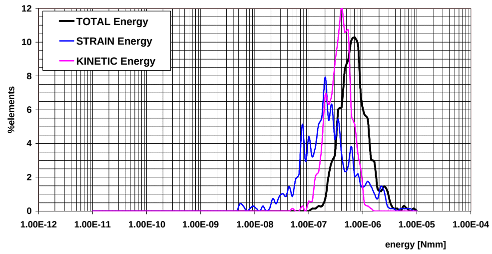

Figure 8 – engine side attachment points: energy distributions and statistics

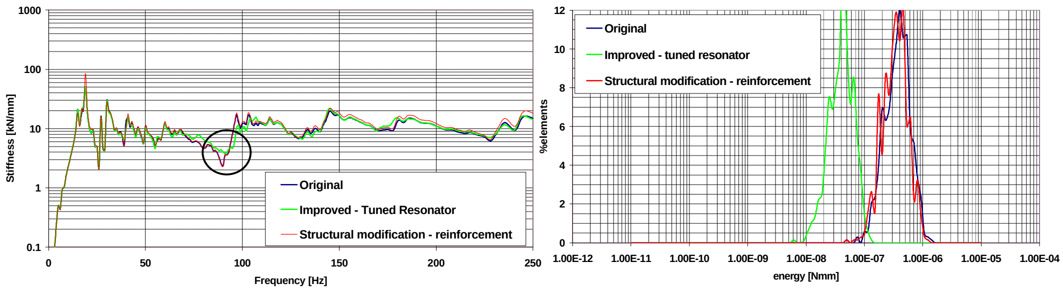

Figure 9 – engine side attachment points: comparison between different improvements

Vehicle vibro-acoustic performance analysis

As a final step, the coupling with the fully-trimmed air cavities has been performed. Special care has been taken in order to simulate the impact of seats, affecting both the absorption inside the passenger cavity and the shape of lower modes. Moreover, investigations have been performed on the role played by the parcel shelf and of the trunk cavity.

The resulting (p / F) were analysed by extracting the modal participation factors for both the structural and fluid parts and by evaluating the panels' contribution: all of them being fundamental data for the acoustical treatment package tuning. As an example, the response for the gearbox-side powertrain attachment was analysed at 90 Hz: the panel participation factors pointed out the front floor and parcel shelf contributions as dominant, with mode-shapes well coupled to the cavity at 126 Hz.

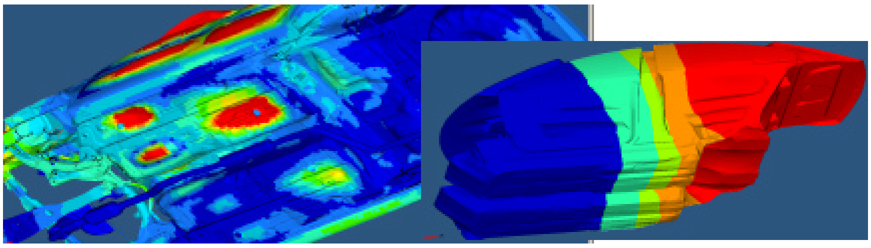

Figure 10 - gearbox attachment points: panel and cavity contributions

CONCLUSIONS

In the present paper, the vibro-acoustic development of a new D-segment vehicle has been described in its main phases, ranging from the "virtual dynamic" model realisation and experimental validation to its optimisation. The simulated inertances (a / F) at the attachments of the vibrating systems together with the (p / F) transfer functions to the passenger cavity have been selected as guiding parameters.

Special focus has been paid to a newly conceived energetic criterion for the structural optimisation phase which, in our opinion, is worth being further developed but already useful for a more systematic selection of interventions in the "localized" structural problems, overcoming the simple trial-and-error approach.

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